Purdue University
Purdue e-Pubs

International Compressor Engineering Conference

School of Mechanical Engineering

1994

Improved Dynamic Valve Analysis for Mass Damping and Moving Frame of Reference

S. J. Schoonmaker
Dresser-Rand

IMPROVED DYNAMIC VALVE ANALYSIS FOR MASS DAMPING AND MOVING FRAME OF REFERENCE

Stephen J. Schoonmaker, ASME, ASQC
Dresser-Rand
Painted Post, New York, USA

ABSTRACT

This paper presents investigations of two special cases of dynamic valve analysis. These special cases arose in the development of a new type of gas field reciprocating compressor. These special cases were for a moving frame of reference (since the discharge valves were integral with the piston), and for independent mass damping (sealing/damping elements are modelled separately instead of as "lumped" masses).

NOMENCLATURE

G	gas damping force	x	distance
S_1	sealing element spring force	KE	kinetic energy
S_2	damping element spring force	D	drag force
m_{p1}	sealing element mass	m_p	mass of element
m_{p2}	damping element mass	V_p	piston speed
y	valve lift	C_d	drag coefficient
g_c	F=ma constant	ρ	gas density
t	time	V_g	gas speed
Ψ	valve lift for fixed coordinates	A_p	plan area of element
y_p	piston position	r	crank radius
a_p	piston acceleration	N	machine speed
p	gas pressure	Z	gas compressibility
R	gas constant	T	gas temperature
V_{sw}	cylinder swept volume	A_{eq}	equivalent flow area

INTRODUCTION

The dynamic valve analysis or "DVA" is used to predict the motion of the moving elements (plates, poppets, etc.) in reciprocating compressor valves. This valve motion prediction is valuable for the proper design and selection of these valves, and this motion prediction represents a significant contribution to reciprocating compressor performance prediction. Computer programs based on Woollatt [1] have proven quite useful in this regard.

In order to assist in the design and analysis of a new type of reciprocating compressor, new demands were placed on the dynamic valve analysis. The first demand was to predict the motion of sealing and damping elements as separate masses. This problem arose since damping elements were failing in field tests, and no acceptable explanation could be determined based on existing dynamic valve analysis methods. The second demand involved predicting the motion of the valve elements (sealing and/or damping plates) in a moving frame reference (since the valve basically comprised the compressor piston in the new design). It was not known what effect this was having on the motion of the damping and sealing plates.

MATHEMATICAL MODEL

Treatment of mass damping

In the case of mass damping, the basic valve or sealing element moves toward full lift, but it is slowed down near the guard by impacting a damping element. Although a lumped mass approach can be used to handle this (Woollatt[2]), that approach does not allow the sealing and damping elements to separate in the damped region, or to re-impact with each other (something that was surmised as causing problems with the new compressor design).

In order to permit the independent motion of the sealing and damping elements, a general numerical method can be used to simulate the differential equation of motion. Specifically, a standard fourth order Runge-Kutta method can be used to solve the initial value problem where the initial gas and valve element conditions are known. As the pressure and forces change, the motion of the elements can be predicted based on changes over a small, finite time step. This is a particularly useful solution given the computing power currently available to engineers.

The following motion possibilities could now be accommodated:

1. Sealing and damping elements at rest.
2. Sealing element in motion below the damping "seat" and the damping element still at rest.
3. Sealing element impacts damping element at the damping "seat".
4. Sealing and damping elements in motion toward the guard.
5. Sealing and damping elements impact the guard together.
6. Sealing element and damping elements leave the guard.
7. Sealing element and damping elements separate during motion toward the seat (prior to reaching the damping "seat").
8. Damping element impacts the damping "seat" (while the sealing element is separated or mated with damping element).
9. Damping element impacts the guard alone.
10. Damping element re-impacts the sealing element.
11. Sealing element impacts the seat (obviously alone).

In each of these cases, adjustments to the motion equation can be made, and/or one can switch from one to two independent motion equations. Note that the model does not allow the sealing and damping elements to separate during motion toward the guard, and that the sealing and damping elements always leave the guard together (they may separate only after the simulation has begun the two elements moving toward the seat).

In order to determine if the elements are separating, the motion equation is evaluated twice. First, damping element lift is predicted based on the sealing and damping elements mated (using one gas force, sealing element's spring force, and damping element's spring force). Secondly, the damping element lift is predicted based on the damping element alone (using no gas force since it is behind the sealing element and damping element's spring force). If the mated case produces a lower damping element lift than the separated case, then the elements are considered to remain mated. Otherwise, the elements are considered to have separated and a motion equation is applied to each element independently. When elements collide, simple conservation of momentum is used to set the element velocities. Impacts with seat or guard and gas damping is handled by previous methods (Woollatt [2]).

Equation of motion for mated sealing and damping elements:

$$\frac{d^2y}{dt^2} = \frac{g_c}{(m_{p1}+m_{p2})}[G-S_1-S_2-D]$$

Equation of motion for damping element alone:

$$\frac{d^2y}{dt^2} = \frac{g_c}{m_{p2}}[-S_2-D]$$

Treatment of moving frame of reference

The new compressor design required modelling the effects of the valve being part of the piston. In this case, the valve element motion (i.e. lift) is relative to the moving piston; thus, the DVA is in a moving frame of reference. Using fixed coordinates, the equation of motion becomes the following:

$$\frac{d^2\Psi}{dt^2} = \frac{g_c}{m_p}[G-S-D]$$

Applying a frame of reference translation, $\Psi = y_p + y$:

$$\frac{d^2y}{dt^2} = \frac{g_c}{m_p}[G-S-D]-\frac{d^2y_p}{dt^2} = \frac{g_c}{m_p}[G-S-D]-a_p$$

As before, the solution of the differential equation is based on a standard numerical method. In this case, the effect of the moving frame of reference is simulated by evaluating the a_p term (based on slider-crank kinematics) for each time step in the simulation.

RESULTS

The mathematical model was incorporated into a new computer program which was verified, documented, and validated according to standard methods (Schoonmaker [3]); this included a verification based on laboratory non-metallic plate motion data presented in Woollatt [2].

Mass damping results

Figure 1. and Table 1. show a typical comparison between the valve motion predicted by the lumped and independent models. As can be seen from the table, there are only small changes in average lift (i.e. valve performance), while there are large changes in the impact speeds (i.e. valve reliability). At this time, it is not clear why there is such a large difference in the guard impact speeds.

FIGURE 1. Valve motion comparison for mass damping models.

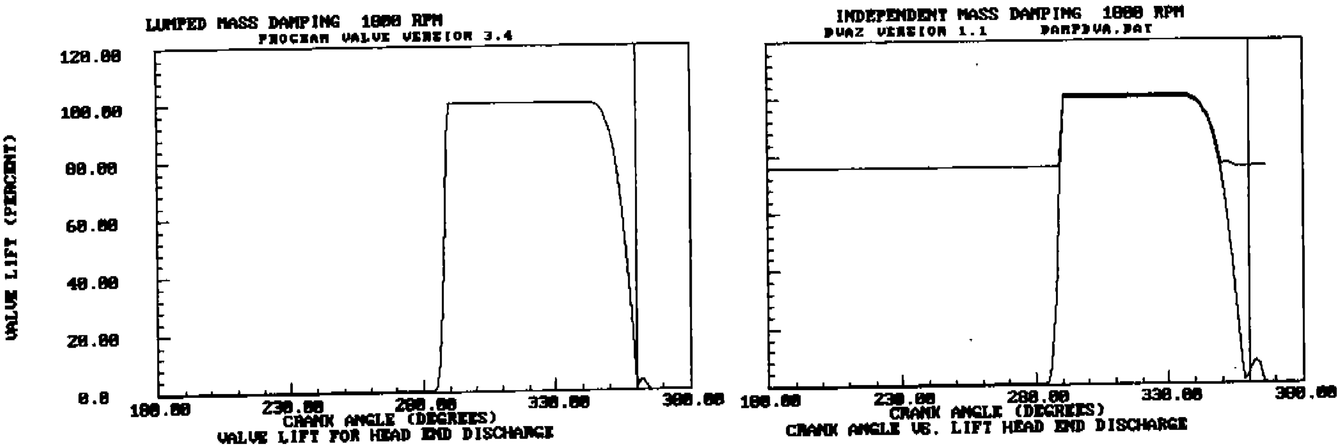


TABLE 1. Model:	Ave. Lift, %	Seat Impact, ft/s (m/s)	Guard Impact	Sealing-Damping Impact	Damping "seat" Impact
Lumped	91.73	9.8 (3.0)	37.4 (11.4)	-	-
Independ.	92.24	7.6 (2.3)	17.8 (5.4)	14.9 (4.5)	3.1 (0.9)

Moving frame of reference results

Figure 2. and Table 2. show a typical comparison between the valve motion predicted with and without the moving frame of reference (mass damping now ignored).

FIGURE 2. Valve motion comparison with and without moving frame of reference.

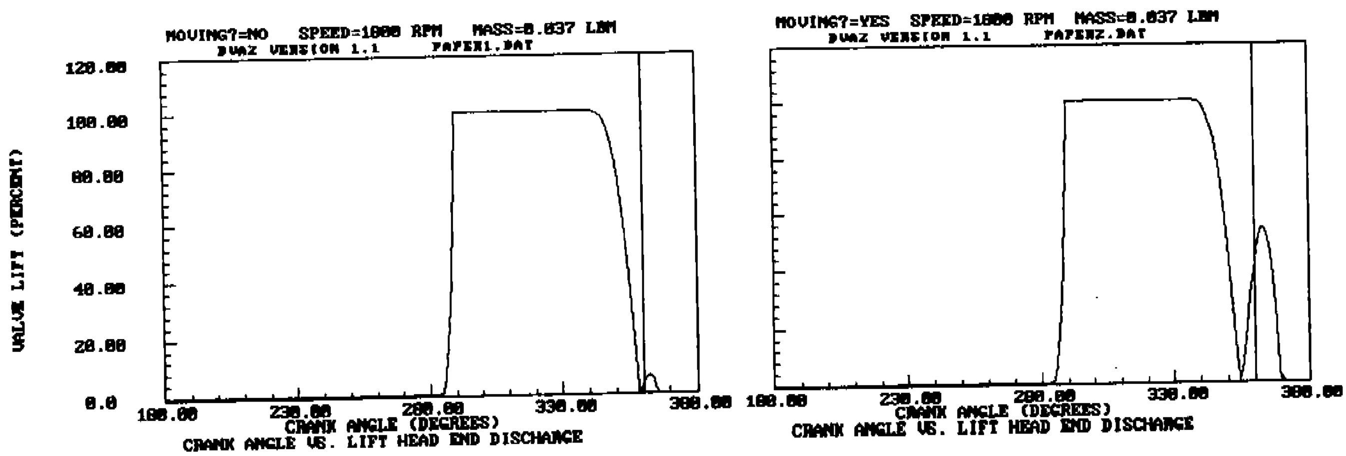


TABLE 2. Moving Frame?	Speed (RPM)	Element mass, lbm (kg)	Max Impact, ft/s (m/s)	Average Lift, %	Change in Ave. Lift, %
No	1800	.037 (.017)	34.8 (10.6)	93.01	-
Yes	1800	.037	34.7 (10.6)	91.45	1.56
No	1800	.100 (.045)	25.1 (7.65)	90.43	-
Yes	1800	.100	25.0 (7.62)	87.07	3.36
No	1800	.250 (.110)	18.4 (5.61)	86.52	-
Yes	1800	.250	18.2 (5.55)	79.53	6.99
No	1200	.037	28.4 (8.66)	89.95	-
Yes	1200	.037	28.3 (8.63)	88.43	1.52
No	1200	.100	20.4 (6.22)	88.72	-
Yes	1200	.100	20.3 (6.19)	84.88	3.84
No	1200	.250	14.9 (4.54)	87.62	-
Yes	1200	.250	14.7 (4.48)	78.92	8.70

The moving frame of reference apparently can adversely affect valve performance (i.e. decreasing the average lift by 3.36% if element mass is 0.100 lb at 1800 RPM). This might be corrected for by a different choice of springs, and in this case, the computer simulation would be essential in assessing the choices. Also, the effect on performance due to changes in element mass appears to be greater than changes in machine speed. For a change in machine speed of 1800 to 1200 RPM (typical values for such a compressor), there is only 14% percent change in the average lift change (3.36 to 3.84 for 0.100 lb). While changing the element mass from 0.037 to 0.100 (typical values for non-metallic and metal plates), there is a 115% change in the average lift change (1.56 to 3.36 for 1800 RPM).

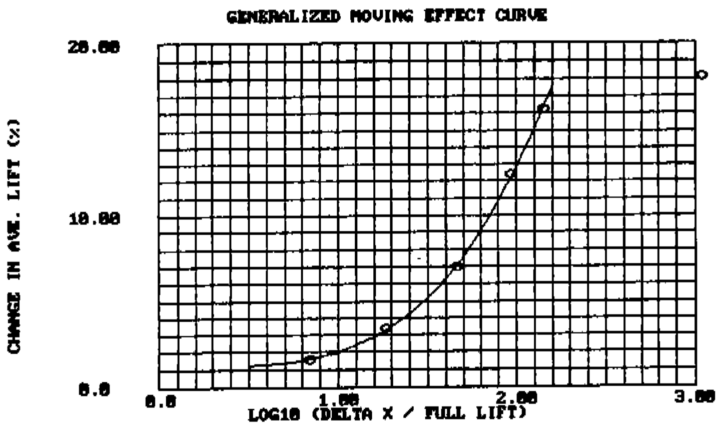
Noting little change in impact speeds, it would appear that the moving frame of reference has little effect on valve reliability.

In order to attempt to predict the effect of the moving frame of reference, one can calculate the distance required to bring the valve element to rest (starting from maximum piston speed) due to a constant drag force alone (using a constant average gas speed). This distance is as follows (note that the machine speed cancels):

$$\Delta x = \frac{\Delta KE}{D} = \frac{\frac{1}{2} m_p V_p^2}{C_d \frac{1}{2} \rho V_g^2 A_p} = \frac{\frac{1}{2} m_p (2\pi N r)^2}{C_d \frac{1}{2} \left(\frac{P}{ZRT}\right) (V_{sw} N / A_{eq})^2 A_p} = \frac{m_p (2\pi r)^2}{C_d \left(\frac{P}{ZRT}\right) (V_{sw} / A_{eq})^2 A_p}$$

This distance can be made dimensionless by dividing by full valve lift. Figure 3. shows the program's prediction of change in average lift due to the moving frame of reference (i.e. the last column in Table 2.) versus the dimensionless distance. This curve could then be used to predict the effect of the moving frame of reference for generalized cases.

FIG. 3. Generalized curve for predicted moving frame of reference effects.



CONCLUSIONS

This paper has presented an investigation of the effect of non-lumped mass damping and a moving frame of reference on the dynamic valve analysis for a new type of reciprocating compressor. The investigation revealed that the non-lumped mass damping model predicts significantly different impact speeds, but little change in valve lifts. The investigation then revealed that the moving frame of reference predicts little change in impact speeds, and there is little change in valve lifts if the mass of the valve moving element is not excessive. A parameter for assessing the effect of moving frame of reference was developed, but it should be researched further (particularly since spring potential energy was ignored). Other future research could include removing the simplification that sealing and damping elements can not separate during motion toward the guard, and attempting to better verify the best mass damping model based on new laboratory and field test data.

REFERENCES

1. Woollatt, D., "A Simple Numerical Solution for Compressor Valves with One Degree of Freedom", Purdue Compressor Conference, 1974.

2. Woollatt, D., and Wertheimer, H., "Improved Accuracy in the Calculation of Valve Dynamics in Reciprocating Compressors", ASME ETCE Conference, January 14-18, 1990, 90-ICE-3.

3. Schoonmaker, S. J., "Engineering Software Quality Management", ASME Petroleum Division - Vol. 43, 1992 Computer Applications and Design Abstractions, pp. 55-63.